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PERFORMANCE OF ASPEN WOOD EXCELSIOR

FOR USE IN EVAPORATIVE COOLERS

By

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Work Unit [illegible]A

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GENERAL AMERICAN RESEARCH DIVISION

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PERFORMANCE OF ASPEN WOOD EXCELSIOR FOR USE IN EVAPORATIVE COOLERS

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GARD Final Report 1288-1 January 1967

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REVIEW NOTICE

This report has been reviewed in the Office of Civil Defense and approved for publication. Approval does not signify that the contents necessarily reflect the views and policies of the Office of Civil Defense.

FOREWORD

The evaporative cooling tests reported herein were conducted by the General American Research Division (GARD) of General American Transportation Corporation, Niles, Illinois under Stanford Research Institute (SRI) Contract No. B64220(4949A-16)-US, OCD Work Unit 1214A. Mr. C. A. Grubb of SRI served as project monitor.

The primary objective of this test program is to determine the performance of aspen wood excelsior when used as the evaporative medium in an evaporative cooler. This performance data can then be used in the design of evaporative coolers for fallout shelters or any other application.

The authors wish to acknowledge the assistance provided during the planning stages of the project by Mr. Robert Ash of the International Metal Products Division of McGraw-Edison Company.

ABSTRACT

Evaporative coolers are devices which have potential application in shelter ventilation and cooling equipment systems analysis. Reported herein are the results of the laboratory tests conducted to determine the saturating effectiveness and air flow resistance of aspen wood excelsior when used as the water evaporating medium in a drip-type evaporative air cooler. The tests were conducted with a cell area of four square feet, cell thicknesses of two and four inches, medium density of 1.5 and 3.0 pounds per cubic foot of cell volume, face velocities ranging from 100 to 500 feet per minute, water flow rates from 0.23 to 0.62 gallon per minute per cubic foot of medium, and inlet air conditions of 80°F dry-bulb temperature (DBT) and 70°F wet-bulb temperature (WBT), 95°F DBT and 76°F WBT, and 90°DBT and 78°WBT.

The saturating effectiveness for the 2-inch thick pad with a density of 1.5 pounds per cubic foot of cell volume decreased steadily from 92 percent at 100 feet per minute face velocity to 73 percent at 500 feet per minute. Doubling the thickness resulted in a saturating effectiveness that was almost independent of the face velocity and constant at 97 percent, while doubling the medium mass density gave a saturating effectiveness that varied from 97 percent at 100 feet per minute to 91 percent at 500 feet per minute. To obtain these saturating effectivenesses, the quantity of water recirculated in the drip-type cooler must exceed five times the amount evaporated into the air.

The air flow resistance of the 2-inch thick pad ranged from 0.017 inch of water at 150 feet per minute to 0.09 inch of water at [illegible] feet per minute. Doubling the thickness of the pad resulted in an air flow resistance which was twice that of the 2-inch pad, while doubling the density increased the air flow resistance by more than three times that of the 2-inch pad.

TABLE OF CONTENTS

SECTION		Page
	FOREWORD	iii
	ABSTRACT	v
1	INTRODUCTION	1
2	THEORY OF EVAPORATIVE COOLING	5
3	TEST APPARATUS AND PROCEDURES	11
	3.1 Test Vehicle	11
	3.2 Air Flow Measuring Station	11
	3.3 Test Chamber	15
	3.3.1 Water Distribution System	15
	3.3.2 Pad Construction	17
	3.3.3 Temperature Measurement	18
4	RESULTS AND DISCUSSION	19
	4.1 Saturating Effectiveness	19
	4.1.1 Saturating Effectiveness for Various Inlet Conditions	20
	4.1.2 Saturating Effectiveness for Various Water Flow Rates	22
	4.1.3 Saturating Effectiveness for an Increased Pad Thickness	22
	4.1.4 Saturating Effectiveness for an Increased Pad Density	24
	4.2 Air Flow Resistance	24
	4.3 Error Analysis	28
	4.4 Conclusions and Application	29
	REFERENCES	32
	BIBLIOGRAPHY	3[illegible]

LIST OF FIGURES

FIGURE NO.		Page
1	Unit-Type Evaporative Coolers	2
2	Heat and Mass Transfer Between Air and Water	[illegible]
3	Typical Evaporative Cooler Process Lines	9
4	Test Apparatus	12
5	Schematic Layout of Test Apparatus	13
6	OCE Test Vehicle	14
7	Exterior View of Test Chamber	16
8	Two-Inch Thick Aspen Wood Evaporating Pad	17
9	Performance Characteristics of a Two-Inch Pad for Variable Inlet Psychrometric Conditions	21
10	Performance Characteristics of a Two-Inch Pad for Variable Water Flow Rates	23
11	Performance Characteristics of a Four-inch Pad	25
12	Performance Characteristics of a 3.0 Pound per Cubic Foot Density Pad	26
13	Air Flow Resistance of Aspen Wood Excelsior Pads	27
14	Performance of Aspen Wood Excelsior Evaporating Pads	30

SECTION 1

INTRODUCTION

Ventilation and cooling devices can be used advantageously in protective shelters; however, the selection of methods for rejecting waste energy from shelters, and the capacity of the equipment required depend upon (Ref. 1) the following parameters:

1. type of shelter (existing buildings, special structures, hardened structures)
2. shelter characteristics (size, interior partitions, apertures)
3. location (weather)
4. occupancy loading
5. allowable heat stress
6. available heat sinks
7. availability of power
8. available equipment
9. cost of the equipment.

Evaporative coolers are devices that have potential application in shelter ventilation and cooling equipment systems analysis. Commercially available evaporative coolers are the drip-type, spray-type, and the rotary pad-type (see Fig. 1). The drip-type evaporative cooler is the most common, and the pads are usually made of aspen wood excelsior. According to the International Metal Products Division of the McGraw-Edison Company (Ref. 2), the commercial units are designed and marketed with [illegible]-inch thick aspen wood pads which have a density of [illegible] pound per square foot. The air system is nominally rated at a pad face velocity of [illegible] feet per minute and a water recirculation rate ten times that being evaporated, thus resulting in a saturating effectiveness of [illegible]

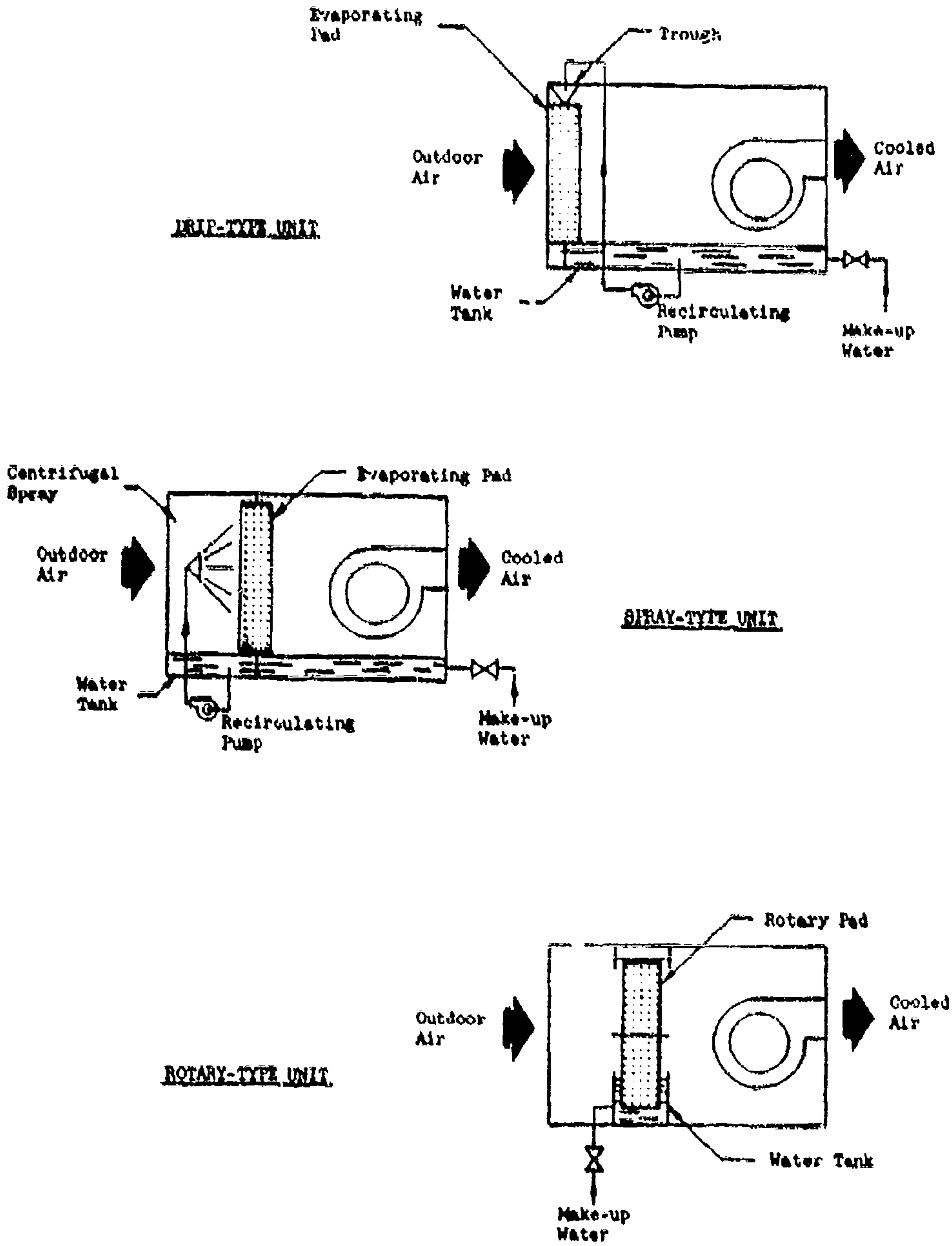


Figure 1 UNIT-TYPE EVAPORATIVE COOLERS

percent. The resistance to air flow for these units is a nominal 0.1 inch water gage.

The spray-type cooler uses metal or glass fibers as the evaporative medium. The saturating effectiveness and pressure drop for some of these media, namely, herringbone-crimped aluminum ribbons, crimped brass wire mesh, nylon fibers, long curled pyrex glass fibers, "Dynel" fibers, and straight glass fibers have been reported (Ref. 3) and are shown in Table I. As can be seen, the saturating effectiveness for 2-inch thick "Dynel" or straight glass fiber pads is greater than 90 percent for face velocities up to 500 feet per minute. Of these, straight glass fibers offer the least resistance to air flow, namely, 0.7 inch water gage at 500 feet per minute. This type of cooler can be designed with a "slinger" in lieu of the nozzle. The saturating effectiveness of the spray-type unit with a "slinger" is roughly 60 to 65 percent because of the non-uniformity of water distribution.

In the rotary-type evaporative cooler, the disk-like pad, which is made of alternate layers of crimped and flat copper or bronze screen, is thoroughly saturated as it rotates through the water. The saturating effectiveness of this device is around 80 to 85 percent; however, maintaining this efficiency is difficult because scaling caused by the use of hard water plugs the mesh screen used for the pads.

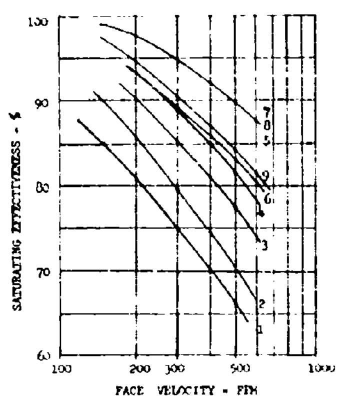
Drip-type evaporative coolers are the simplest of the three types to put into operation and use aspen wood as an evaporative medium. Since little performance data is available for aspen wood excelsior, this investigation was initiated because it appeared that the resistance to air flow by aspen wood is considerably less than that for any other medium. In addition, aspen wood excelsior is the least expensive medium available for use in evaporative coolers.

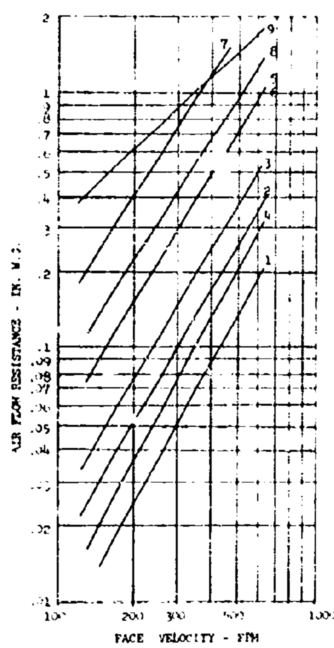
TABLE I

PERFORMANCE CHARACTERISTICS OF WET CELL AIR HUMIDIFIERS

CURVE NO.	CELL	DESCRIPTION	THICKNESS (in.)	PACKING SURFACE (sq ft/cu ft)
1			2	155
2	Crimped Metal Ribbon	Packed with herringbone-crimped aluminum ribbons 0.009 in. thick, 12 in. long	4	155
3			6	155
4	Crimped Metal Mesh	Multiple layers of herringbone-crimped brass wire mesh woven from 0.01 in. diam wire and having 16 openings per in.	4	120
5	Straight Glass Fibers	4-8 in. long glass fibers, 135 micron diam. random-packed in planes parallel to cell face	2	[illegible]
6	Crimped Nylon Fibers	4 in. long crimped nylon fibers, 250 micron diam, random-packed in planes parallel to cell face	1.75	81
7	6 Denier Dynel Fibers	Bonded fiber pack containing 0.5 lb of 25 micron diam fibers per cu ft of cell volume (63% fibers and 27% bonding agent by wt)	2	[illegible]
8	24 Denier Dynel Fibers	Bonded fiber pack containing 0.6 lb of 48 micron diam fibers per cu ft of cell volume (63% fibers and 27% bonding agent by wt)	2	[illegible]
9	Curly Glass Fibers	1-2 in. long curled pyrex glass fibers 37 micron diam.	2	168

NOTE: Using 2-1/4" x 3/32" Hollow Cone Nozzles, Water Rate = 0.6 gpm/sq ft cell face area





SECTION 2

THEORY OF EVAPORATIVE COOLING

Evaporative cooling is a process during which air is both cooled and humidified by bringing it into contact with water. The combined process of heat and mass transfer that occurs as the air passes over a water film (see Figure 2) can be expressed quantitatively

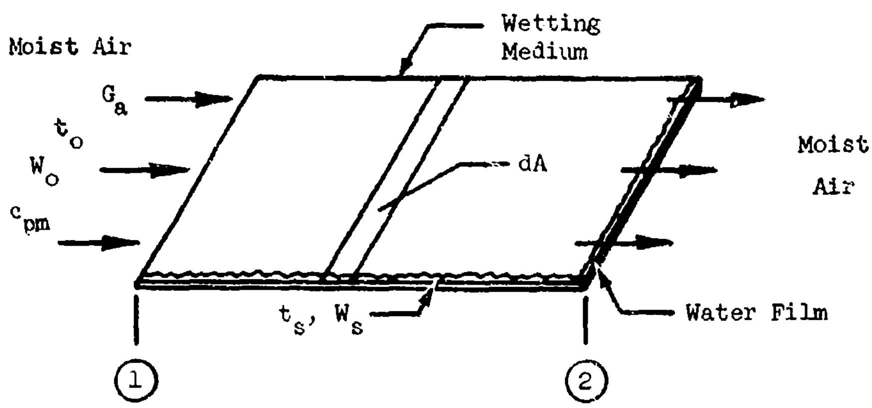


Figure 2 HEAT AND MASS TRANSFER BETWEEN AIR AND WATER

for the differential area, dA, as

$$dq_s = h_c \, dA \, (t_s - t_o) \qquad (1)$$

and

$$dm_s = k \, dA \, (W_s - W_o) \qquad (2)$$

where:

dq_s = rate of sensible heat transferred, Btu/hr

h_c = convection heat transfer coefficient, Btu/hr-sq ft - °F

dm_s = rate of moisture transferred, lb/hr

k = diffusion constant, lb water/(lb water per lb dry air)-sq ft-hr

G_a = mass flow rate of moist air, lb/hr

dA = differential area of contact between air and water, sq ft

t_s = temperature of the water film, °F

t_o = dry-bulb temperature of air, °F

W_s = humidity ratio of saturated air at the temperature of the water film, lb water/lb dry air

W_o = humidity ratio of air, lb water/lb dry air

c_{pm} = specific heat of moist air at constant pressure, Btu/lb-°F.

For an adiabatic process, the sensible heat transferred is equal to the corresponding change in dry-bulb temperature of the air; therefore,

$$dq_s = G_a c_{pm} \, dt_o \tag{3}$$

Since all of the water evaporated goes entirely to raise the humidity ratio of the air, then

$$dm_s = G_a \, dW_o \tag{4}$$

Rewriting Equations 1, 2, 3 and 4, the following expressions result:

$$h_c \, dA \, (t_s - t_o) = G_a \, c_{pm} \, dt_o \tag{5}$$

$$k \, dA \, (W_s - W_o) = G_a \, dW_o \tag{6}$$

Integrating Equations 5 and 6 over area A, between points 1 and 2, (see Figure 2), and holding h_c, k, t_s, and W_s constant yields

$$\frac{h_c}{G_a \, c_{pm}} \int_{0}^{A} dA = \int_{t_1}^{t_2} \frac{dt_o}{t_s - t_o}$$

or

$$\frac{h_c A}{G_a \, c_{pm}} = - \ln \left[\frac{t_s - t_2}{t_s - t_1} \right] \tag{7}$$

$$\frac{k}{G_a}\int_0^A dA = \int_{W_1}^{W_2} \frac{dW_o}{W_s - W_o}$$

$$\text{or} \quad \frac{kA}{G_a} = -\ln\left[\frac{W_s - W_2}{W_s - W_1}\right] \tag{8}$$

Either equation 7 or 8 may be used to define the saturating effectiveness. Since temperatures are more easily measured experimentally than humidity ratios, equation 7 is preferred and can be rewritten as

$$\left[\frac{t_s - t_2}{t_s - t_1}\right] = \exp\left(-h_c A/G_a c_{pm}\right) \tag{9}$$

and the terms rearranged to give:

$$\left[\frac{t_1 - t_2}{t_1 - t_s}\right] = 1 - \exp\left(-h_c A/G_a c_{pm}\right) \tag{10}$$

Since the mass flow rate of air, G_a, can be expressed in volumetric terms as

$$G_a = 60\rho A_p V \tag{11}$$

equation 10 can be rewritten as

$$E = \left[\frac{t_1 - t_2}{t_1 - t_s}\right] = 1 - \exp\left(-h_c A/60\rho A_p V c_{pm}\right) \tag{12}$$

where:

G_a = mass flow rate of moist air, lb/hr

ρ = density of inlet air, lb/cu ft

A_p = face area of the pad, sq ft

V = face velocity of the air entering the pad, ft/hr

t_1 = inlet air dry-bulb temperature, °F

t_2 = outlet air dry-bulb temperature, °F

t_s = temperature of the water film, °F.

The ratio of temperatures in equation 12 is defined as the saturating effectiveness, E, of an adiabatic saturator.

The adiabatic saturating process is one during which the total energy of the air remains constant, and therefore can be represented on the psychrometric chart by a constant enthalpy line. However, due to the small amount of energy which is added to the air stream by the liquid evaporated, the process is usually represented by a constant wet-bulb temperature line (see Figure 3). In reality, evaporative coolers are not adiabatic devices due to the heat exchange through the cooler cabinet, the energy added to the water by the recirculating pump, and the temperature of the make-up water; hence, the wet-bulb temperature of the air does not remain constant. Process 1-2' as shown in Figure 3 represents a non-adiabatic device in which energy is rejected from the system, and process line 1-2" is for a system to which energy is added. The saturating effectiveness for an actual evaporative cooler is defined as (Reference 3)

$$E_{act} = 1 - \left[\frac{t_2 - t_{w2}}{t_1 - t_{w1}}\right] \tag{13}$$

where:

E_{act} = saturating effectiveness, dimensionless

t_2 = dry-bulb temperature of outlet air, °F

t_1 = dry-bulb temperature of inlet air, °F

t_{w2} = wet-bulb temperature of outlet air, °F

t_{w1} = wet-bulb temperature of inlet air, °F.

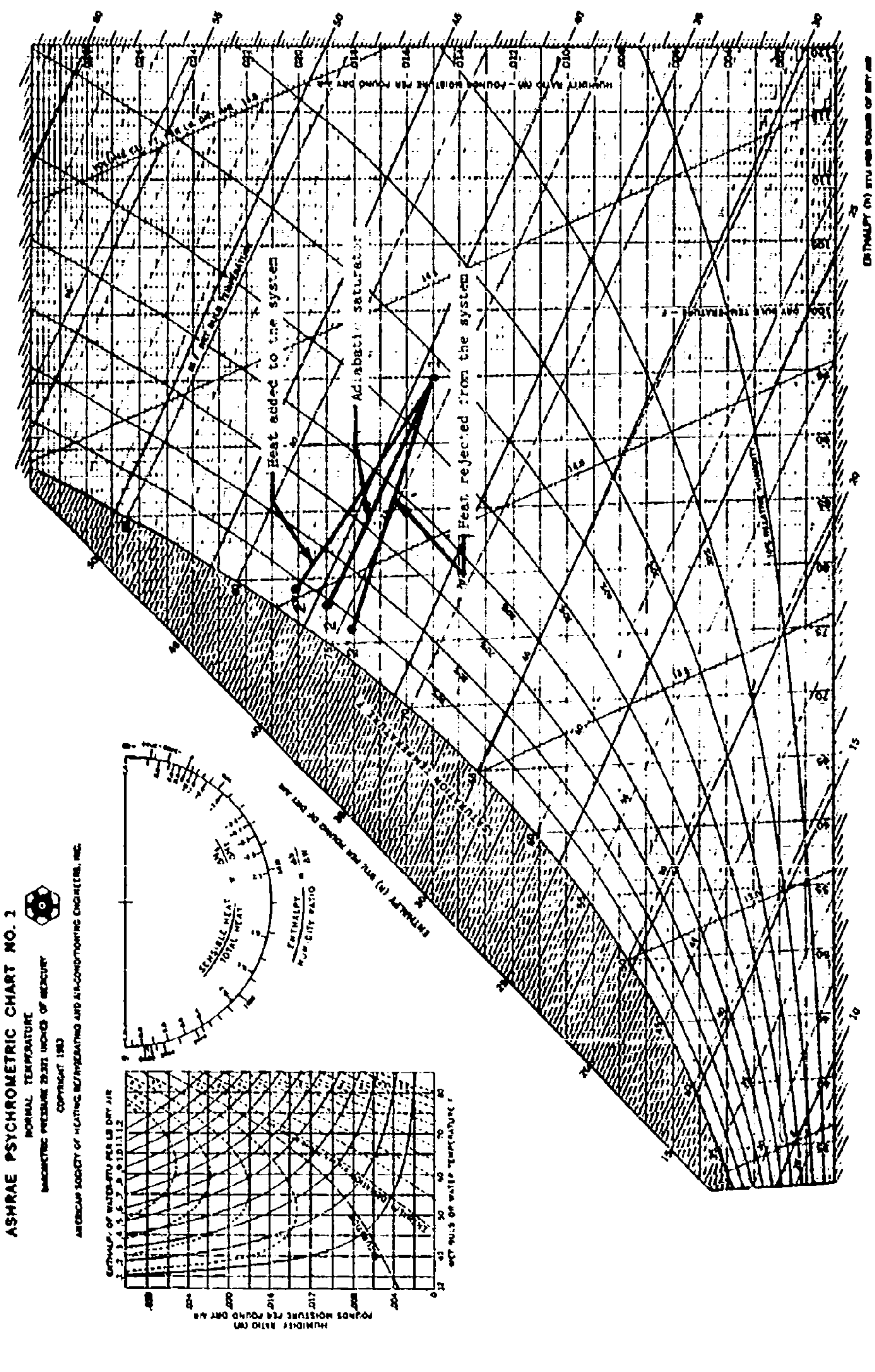


Figure 3 TYPICAL EVAPORATIVE COOLER PROCESS LINES

The test apparatus as described in the following section was designed so that the face velocity, inlet air dry and wet-bulb temperatures, water flow through the evaporating pads, the pad thickness, and the pad density could be varied independently from each other. These controllable (independent) variables affect the parameters (dependent variables) in equation 12 as shown in Table II.

TABLE II

SUMMARY OF TEST VARIABLES AFFECTING SATURATION EFFECTIVENESS

Controllable Test Value	Dependent Variable (see Equation 12)
Inlet Air Conditions	ρ and c_{pm}
Air Flow Rate	V
Water Flow Rate	h_c and A
Pad Thickness and Density	h_c and A

SECTION 3

TEST APPARATUS AND PROCEDURES

The apparatus consisted of a test vehicle, a 30-foot air flow measuring station, and a test chamber as shown in Figures 4 and 5. With this arrangement, the face velocity through the evaporating pads, the inlet air dry and wet-bulb temperatures, the density and thickness of the pads, and the water flow to the pads could be controlled and varied.

3.1 Test Vehicle

The air supply for the test chamber was obtained from the Office of Civil Defense (OCD) Test Vehicle No. 1 (Figure 6, References 4 and 5). The Test Vehicle is capable of supplying up to 8,600 cfm of air at a predetermined dew point temperature and a dry-bulb temperature. Dehumidification is accomplished by a 20-ton water chiller, and humidification and reheat are both achieved by a hot water boiler that has a gross output of 800,000 Btu per hour. After the dew point temperature is obtained, the dry-bulb temperature is controlled by reheating the supply air as it passes through a hot water coil. The air flow is manually controlled by adjusting the fan speed, the fan inlet vanes and the air by-pass ports. The Test Vehicle was connected to the air flow measuring station by a 24-inch diameter, flexible, wire-reinforced cloth duct.

3.2 Air Flow Measuring Station

The design of the air flow measuring station was based on recommendations of the National Electrical Manufacturers Association (Reference 6), and was fabricated with rigid 12-inch diameter spiral conduit duct. Air flow rates were measured with an 8-inch diameter aperture, sharp-edge, orifice plate across which the differential static pressure was measured with a manometer

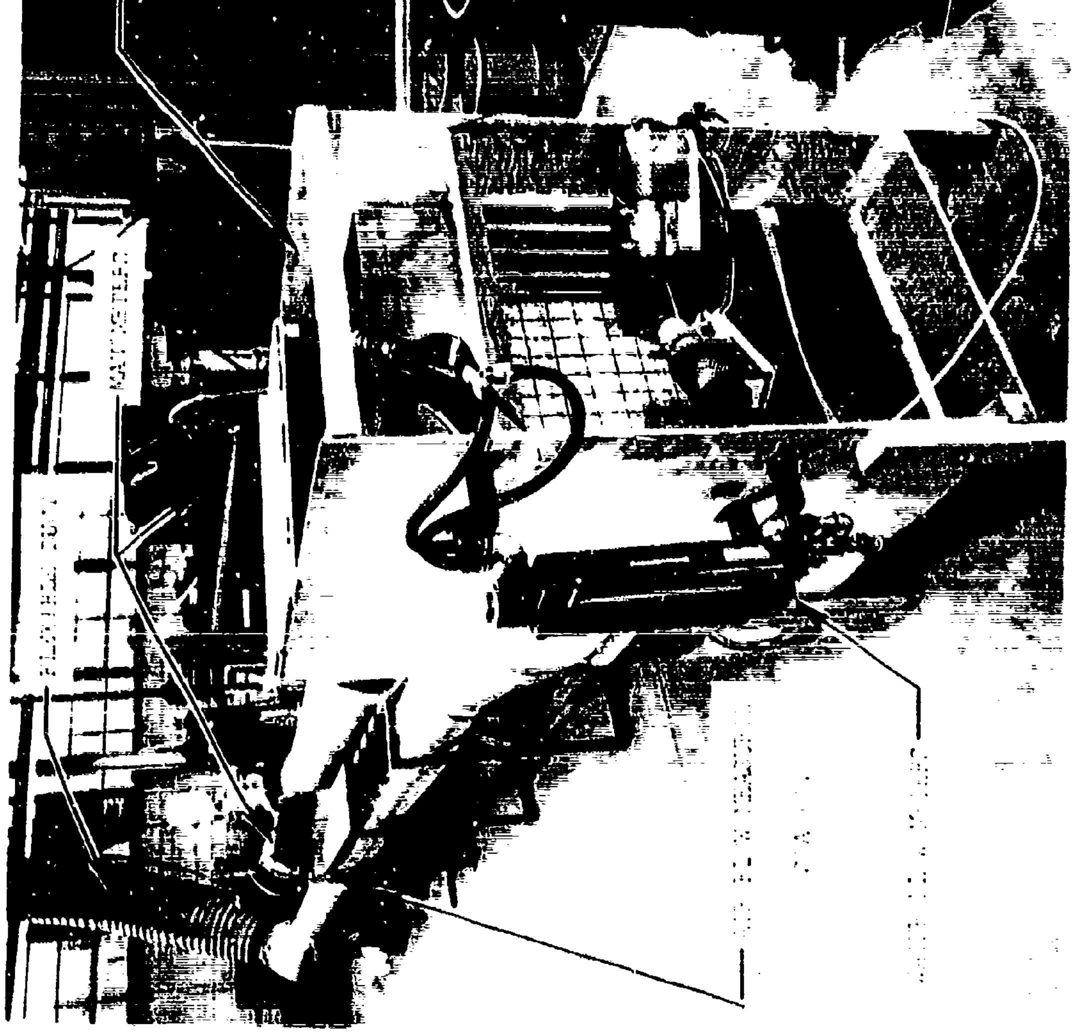


Figure 4 TEST APPARATUS

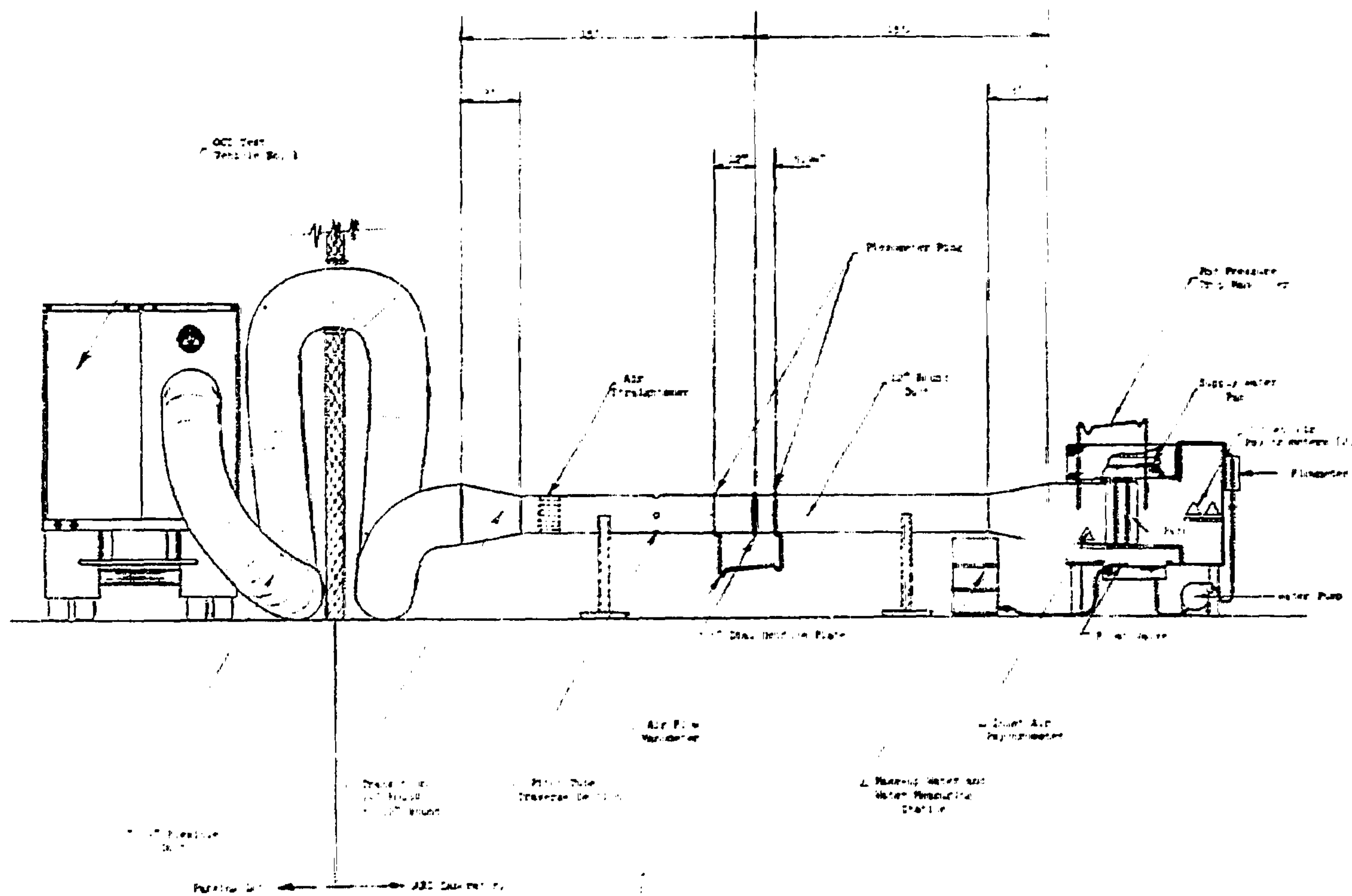


Figure 5 SCHEMATIC LAYOUT OF TEST APPARATUS

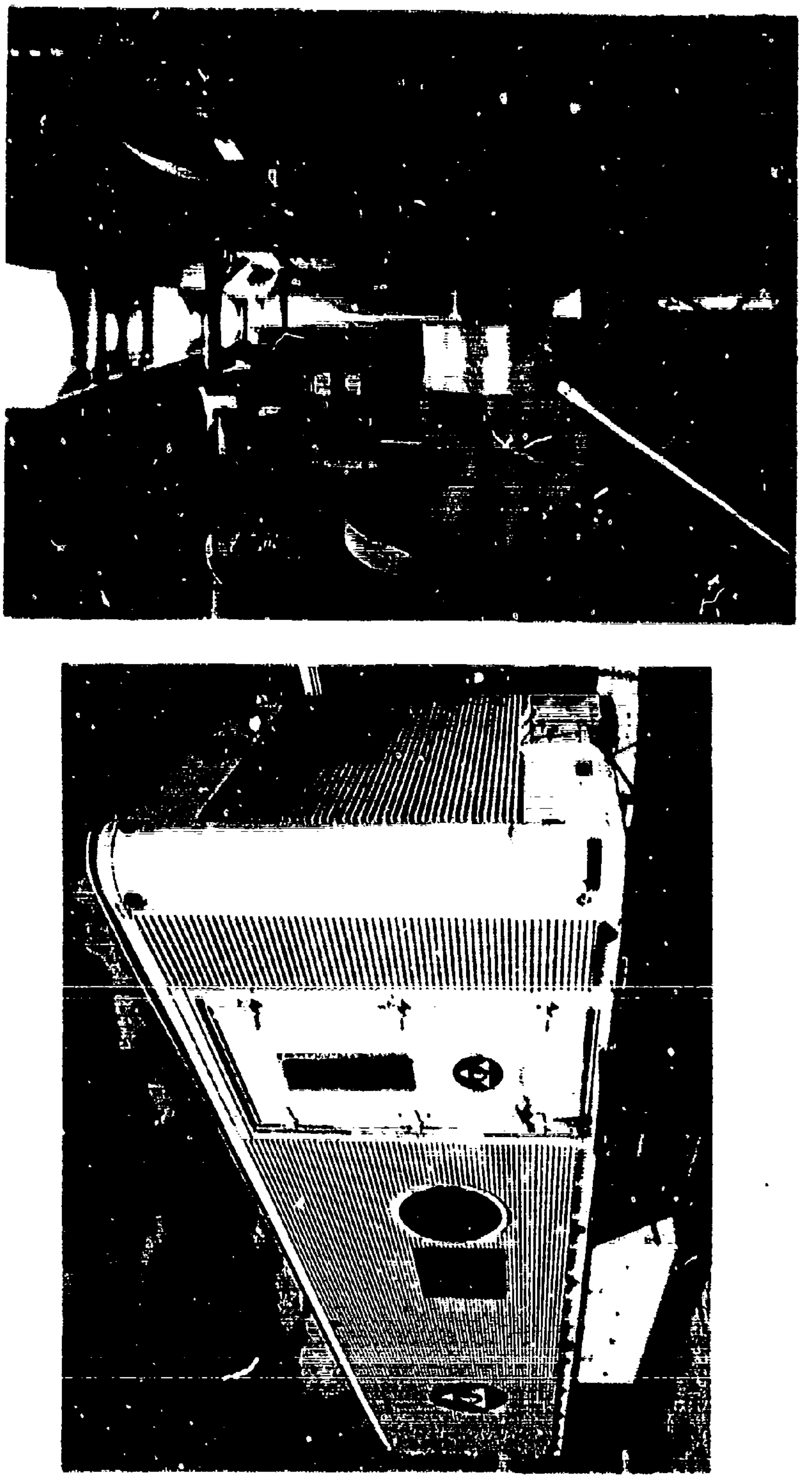


Figure 6 OCD TEST VEHICLE

connected to piezometer rings as shown in Figure 5. The orifice plate was calibrated with a pitot tube which was located four feet upstream from the orifice. The barometric pressure and dry and wet-bulb temperatures were recorded before each test to provide for correction of the air flow to standard air (0.075 pound per cubic foot).

3.3 Test Chamber

The test chamber (see Figure 7) measured 4 feet by 4 feet by 2 feet and was constructed of 3/4-inch plywood to minimize heat exchange with the surroundings. The chamber housed the water distribution system, pads, and temperature measuring equipment. Air entered the chamber through a 12-inch diameter duct. To ensure that the face velocity entering the pad was uniform, the air was passed through a fine mesh screen. The variation in face velocity entering the pad did not exceed ± 15 percent.

3.3.1 Water Distribution System

Water was distributed over the pads from the supply pan which had five rows of 12 holes that were of 1/16-inch diameter and spaced on 2-inch centers. A visual inspection indicated that the entire pad was wetted with this arrangement. The water that was not evaporated as it passed through the pads was collected in a sump below the pads where a pump recirculated the water through a filter and back to the supply pan. Make-up water at room temperature replaced the water which was evaporated. The rate of water flow over the pads was controlled by an adjustable float valve that maintained a constant water level in the supply pan. When the level of water in the pan was constant, the reading indicated on the flowmeter equalled the amount of water flowing over the pads. This arrangement allowed very accurate control of the water flow in a range from 0.15 to 2.75 gallons per minute. The amount of water evaporated or make-up water

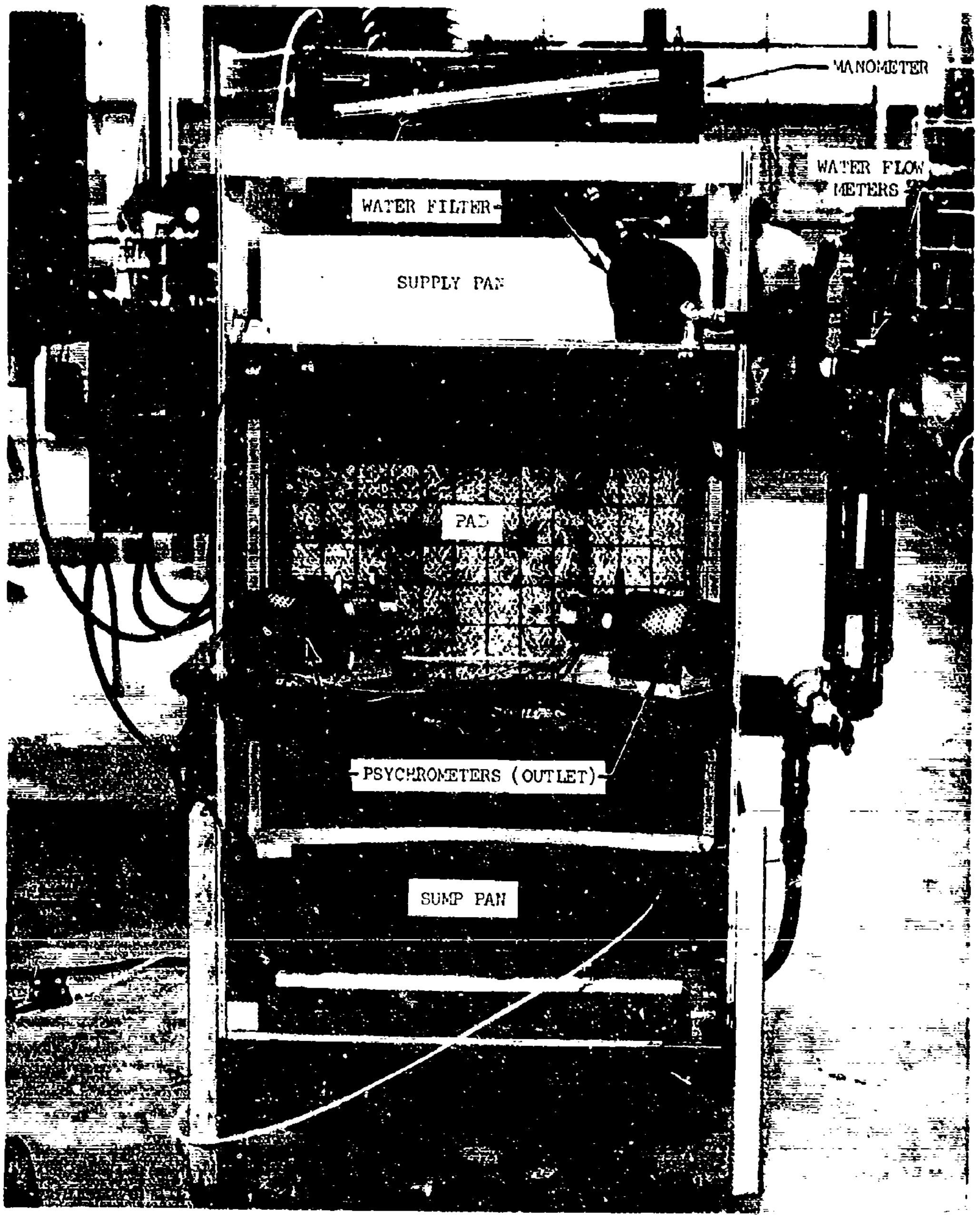


Figure 7 EXTERIOR VIEW OF TEST CHAMBER

required is easily determined for any inlet air condition once the air flow rate and saturating effectiveness are known.

3.3.2 Pad Construction

Aspen wood excelsior was used to fabricate 2-inch thick pads that had a face area of four square feet and a density of 1.5 and 3.0 pounds per cubic foot of cell volume. A cloth netting and metal screen (see Figure 8) were placed around the excelsior to prevent wood fibers from falling into the sump pan and to give the pad rigidity against the force of the air blowing through it. The test chamber was constructed so that pads could be placed in series to simulate pad thicknesses of two and four inches. The excelsior was packed on a light table so that good uniformness of density was achieved.

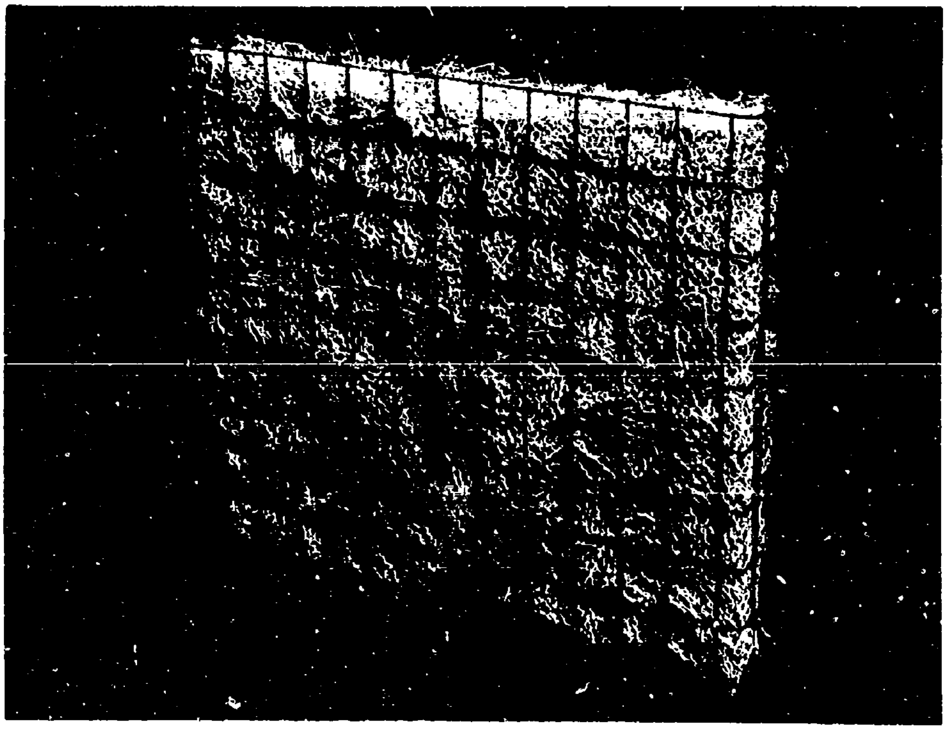


Figure 8 TWO-INCH THICK ASPEN WOOD EVAPORATING PAD

3.3.3 Temperature Measurement

Inlet and outlet air temperatures were measured with aspirating psychrometers (Sargent Model S-42010) which were equipped with resistance thermometers (Minco No. 1119). One psychrometer was placed upstream of the pad to measure the dry and wet-bulb temperature of the incoming air, and two psychrometers were located at the outlet of the test chamber to measure the conditions of the air leaving the pad. In addition, resistance thermometers were located in the supply pan and sump pan to measure the temperature of the water entering and leaving the pad.

SECTION 4

RESULTS AND DISCUSSION

4.1 Saturating Effectiveness

Nine series of tests were conducted to determine the effect of varying face velocities, water supply rates, inlet air conditions, pad thicknesses and pad densities on the saturating effectiveness of aspen wood excelsior. The test conditions are summarized in Table III.

TABLE III

SUMMARY OF TESTS

Test Series No.	Inlet Air		Pad Thickness, inches	Water Flow Rate, gal/min/cu ft of cell volume	Pad Density, lbs/cu ft of cell volume	Purpose of Test
	*DBT, °F	†WBT, °F				
1	80	70	2	0.48	1.5	Determine effect of inlet air temperatures on performance
2	95	76	2	0.48	1.5	
3	90	78	2	0.48	1.5	
4	90	78	2	0.23	1.5	Determine effect of water flow rate on performance
5	90	78	2	0.37	1.5	
6	90	78	2	0.62	1.5	
7	90	78	4	0.23	1.5	Determine effect of doubling the pad thickness
8	90	78	4	0.37	1.5	
9	90	78	2	0.37	3.0	Determine effect of doubling the pad density

*Dry Bulb Temperature
†Wet Bulb Temperature

4.1.1 Saturating Effectiveness for Various Inlet Conditions

The parameters appearing in equation 12 that are affected by the inlet air conditions are the air density, ρ , and the specific heat of moist air, c_{pm}. Air density, however, varies little over the temperature range encountered in air cooling work, and therefore it can be considered as a constant. The value of ρ used most often for engineering calculations is that of standard air, 0.075 lb/cu.ft. The specific heat of moist air, c_{pm}, is defined by the equation

$$c_{pm} = 0.24 + 0.45\, W \tag{14}$$

where W is the humidity ratio expressed in units of lb water/lb dry air. Although the humidity ratio varies considerably (0.010 to 0.020 lb water/ lb dry air) for the temperature range of concern, the product 0.45 W is small (2 to 4 percent) compared to 0.24. The specific heat of moist air, therefore, can also be considered a constant -- the usual accepted value of c_{pm} being 0.245 Btu/lb - °F.

Since ρ and c_{pm} are the only variables in equation 12 affected by the inlet air conditions, and since they can be considered constant over the temperature range encountered, the assumption is thus made that the saturating effectiveness, E, is independent of the inlet air conditions. Test Series No. 1, 2 and 3 were run to determine the validity of this assumption. These tests were performed with 2-inch thick pads, 1.5 pounds of aspen wood per cubic foot of cell volume, 0.48 gpm of water per cubic foot of aspen wood, face velocities ranging from 100 to 600 feet per minute (fpm), and inlet conditions as scheduled. The results of these tests are shown in Figure 9, where a linear least squares regression line was fitted to the data. The heavy vertical lines which extend through each data point represent the possible variance of E due to experimental inaccuracy (see Section 4.3). Even though another correlation of the data might be found,

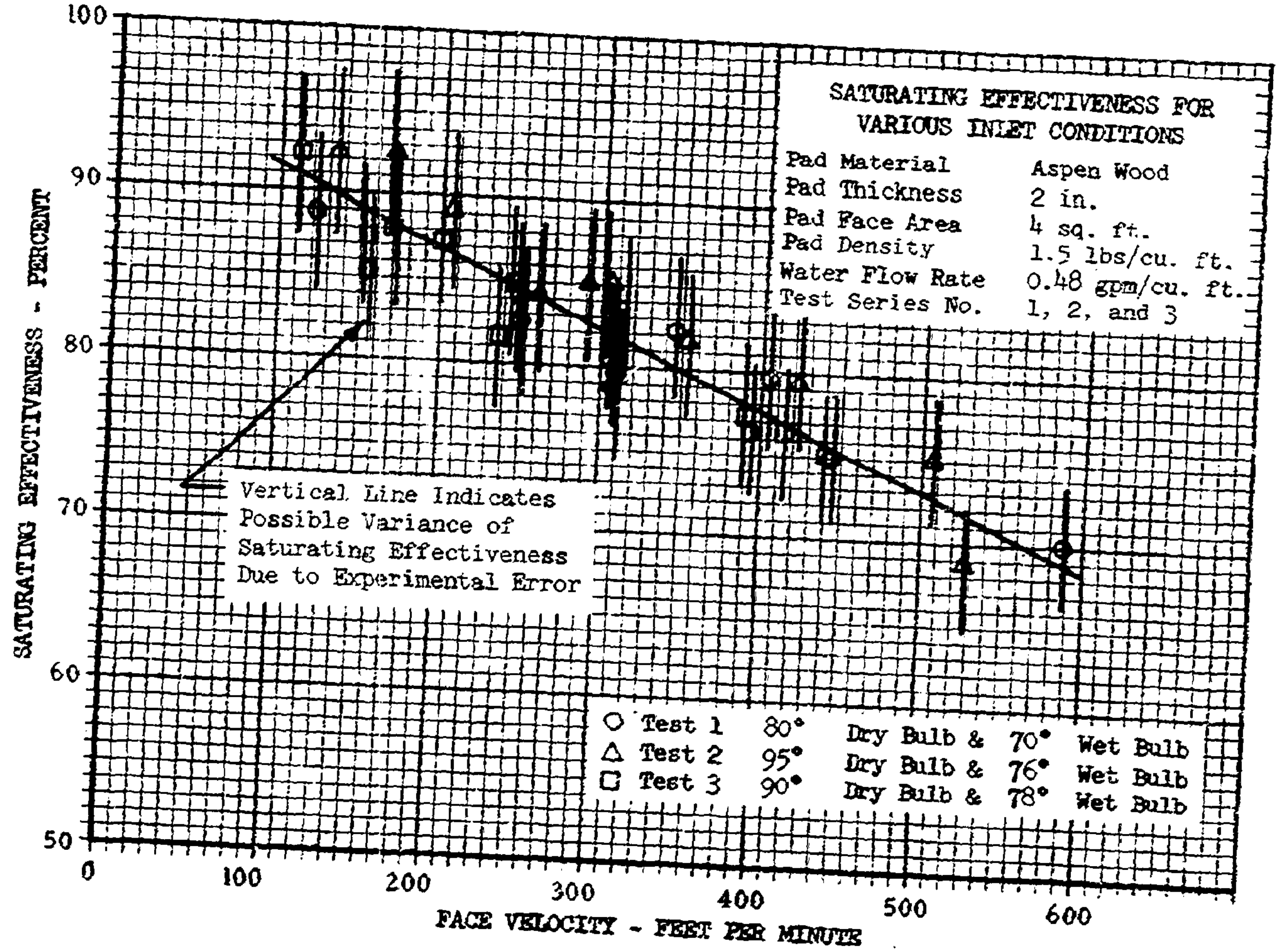


Figure 9 PERFORMANCE CHARACTERISTICS OF A TWO-INCH PAD FOR VARIABLE INLET PSYCHROMETRIC CONDITIONS

the fact is illustrated in Figure 9 that the data can be fitted with a single correlation and hence the experimental results substantiate the assumption that the saturating effectiveness can be considered to be independent of the inlet air conditions for the temperature range of concern. All remaining tests were run with an inlet air condition of 90°F dry-bulb and 75°F wet-bulb temperatures.

4.1.2 Saturating Effectiveness for Various Water Flow Rates

Test Series No. 4, 5 and 6 were performed in addition to Test Series No. 1, 2 and 3 to determine the effect of the water supply rate on the saturating effectiveness. The water supply rates were varied from 0.23 to 0.62 gallon per minute per cubic foot of aspen wood, and the results are presented in Figure 10. The curve, which was fitted with a least squares regression line, suggests that the saturating effectiveness is independent of the water supply rate, since the data can again be fitted with a single correlation and indeed, the least squares regression line is almost identical to that which resulted in Section 4.1.1.

Since the minimum water rate recirculated over the pad was 0.23 gallon per minute per cubic foot of aspen wood, or five times the quantity being evaporated, it is cautioned that the results reported herein not be extrapolated to lower water rates. It is recommended that the pump selected should be capable of recirculating water at five times the evaporation rate. For a 2-inch thick aspen wood pad and a water rate approximately five times that being evaporated, the saturating effectiveness is 92 and 73 percent at face velocities of 100 and 500 fpm, respectively.

4.1.3 Saturating Effectiveness for an Increased Pad Thickness

As the thickness of the pad is increased, the total effective wetted area of contact between the air and water is increased. Test Series 7 and 8 were conducted to determine the effect of doubling the pad thickness. The pad density for

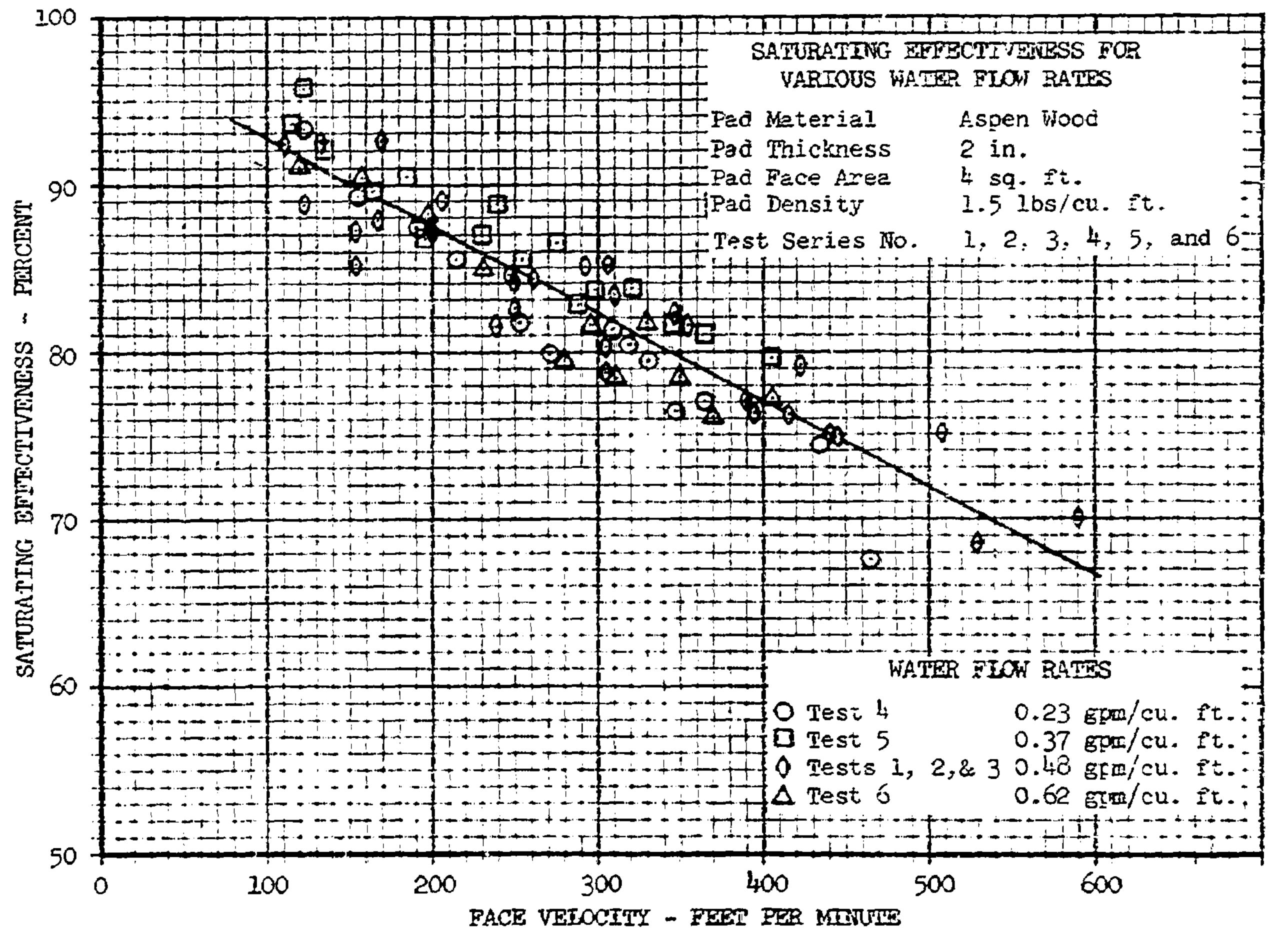


Figure 10 PERFORMANCE CHARACTERISTICS OF A TWO-INCH PAD FOR VARIABLE WATER FLOW RATES

these tests was 1.5 pounds per cubic foot, and the water flow rates were 0.23 and 0.37 gallon per minute per cubic foot. As can be seen from Figure 11, the saturating effectiveness is increased considerably and is less dependent upon the face velocity than the 2-inch thick pad. At face velocities of 100 fpm and 450 fpm, the saturating effectivenesses are approximately 97 and 94 percent, respectively.

4.1.4 Saturating Effectiveness for an Increased Pad Density

For a 2-inch thick pad with a water flow rate of 0.37 gpm per cubic foot, the effect of increasing the pad density from 1.5 to 3.0 pounds per cubic foot resulted in an increase of the saturating effectiveness from 73 to 92 percent at a face velocity of 450 fpm (see Figure 12). The increased saturating effectiveness is primarily a result of the increased total effective wetted area of contact between the aspen wood and the air.

4.2 Air Flow Resistance

During each test, the pressure drop of the pads was measured and the results are presented in Figure 13. All pads during these tests were wet. As the pad thickness, the water flow rate through the pads, the pad density and the face velocity are increased, the resistance to air flow is increased. For a 2-inch thick pad, the air flow resistance is increased from 0.017 inch of water (iwg) at 150 fpm to 0.108 iwg at 450 fpm for water flow rates ranging from 0.23 to 0.37 gpm per cubic foot of media. When doubling the density of the pad, the pressure drop at 300 fpm is increased from 0.058 iwg to 0.211 iwg.

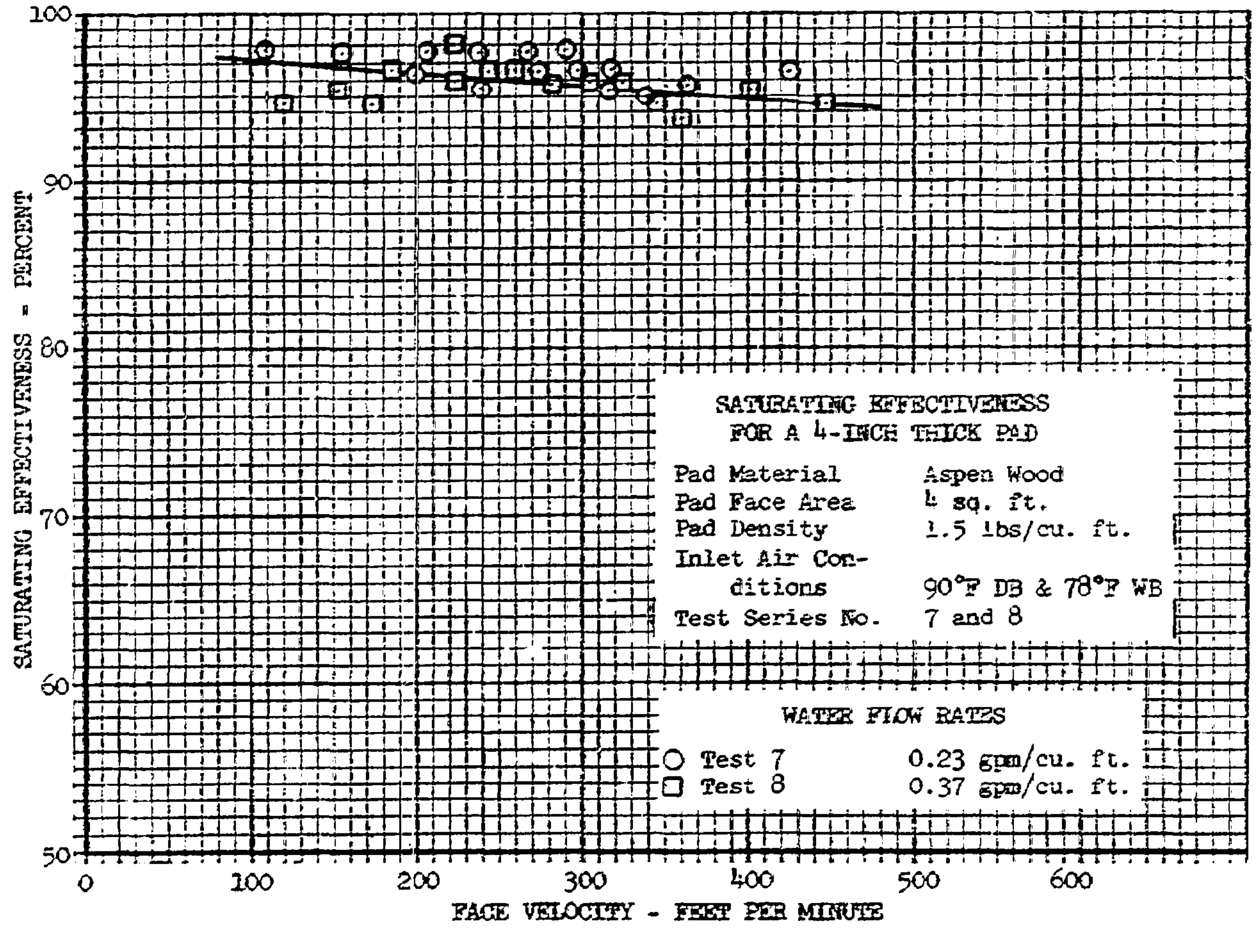


Figure 11 PERFORMANCE CHARACTERISTICS OF A FOUR-INCH PAD

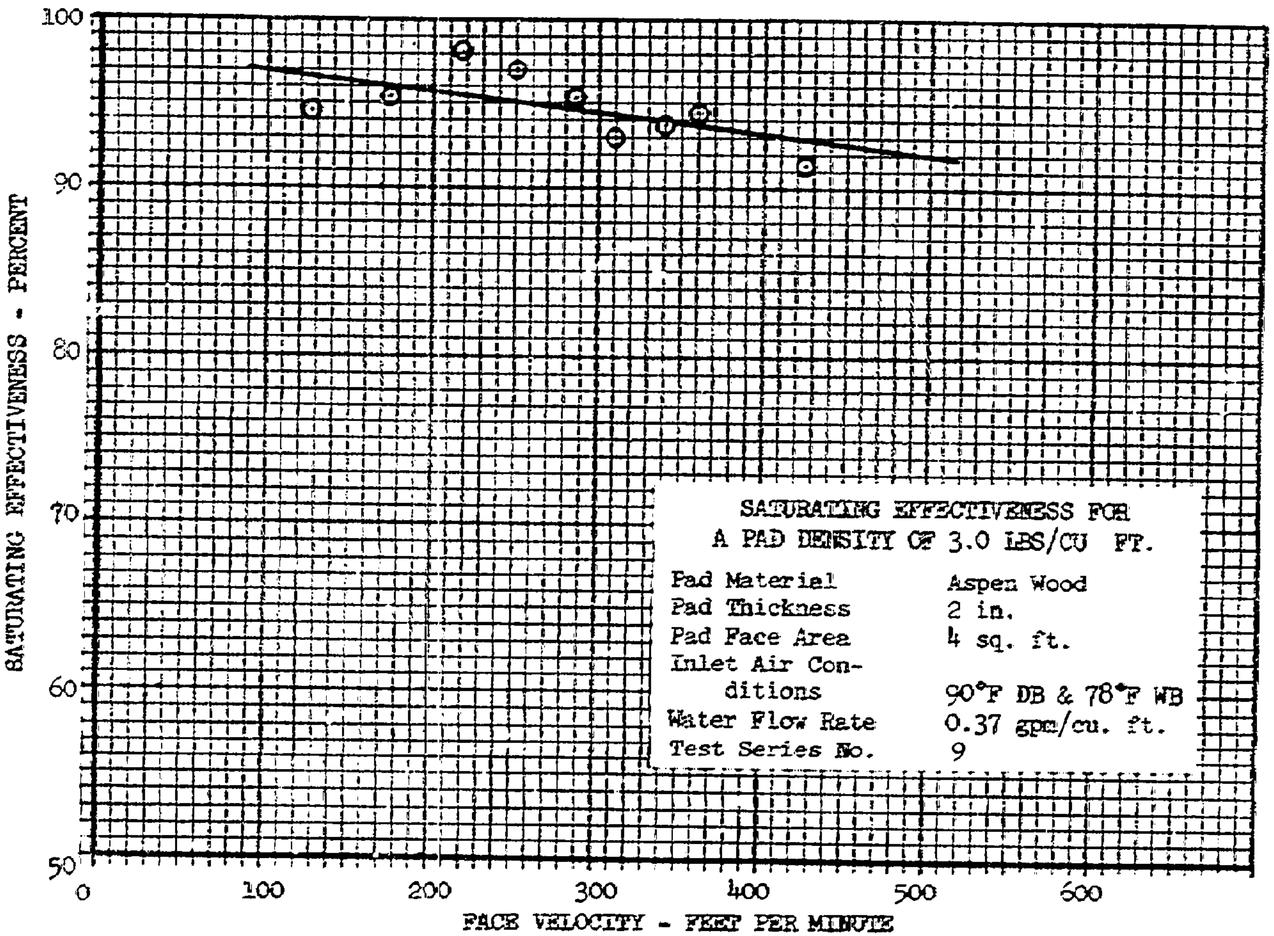


Figure 12 PERFORMANCE CHARACTERISTICS OF A 3.0 POUND PER CUBIC FOOT DENSITY PAD

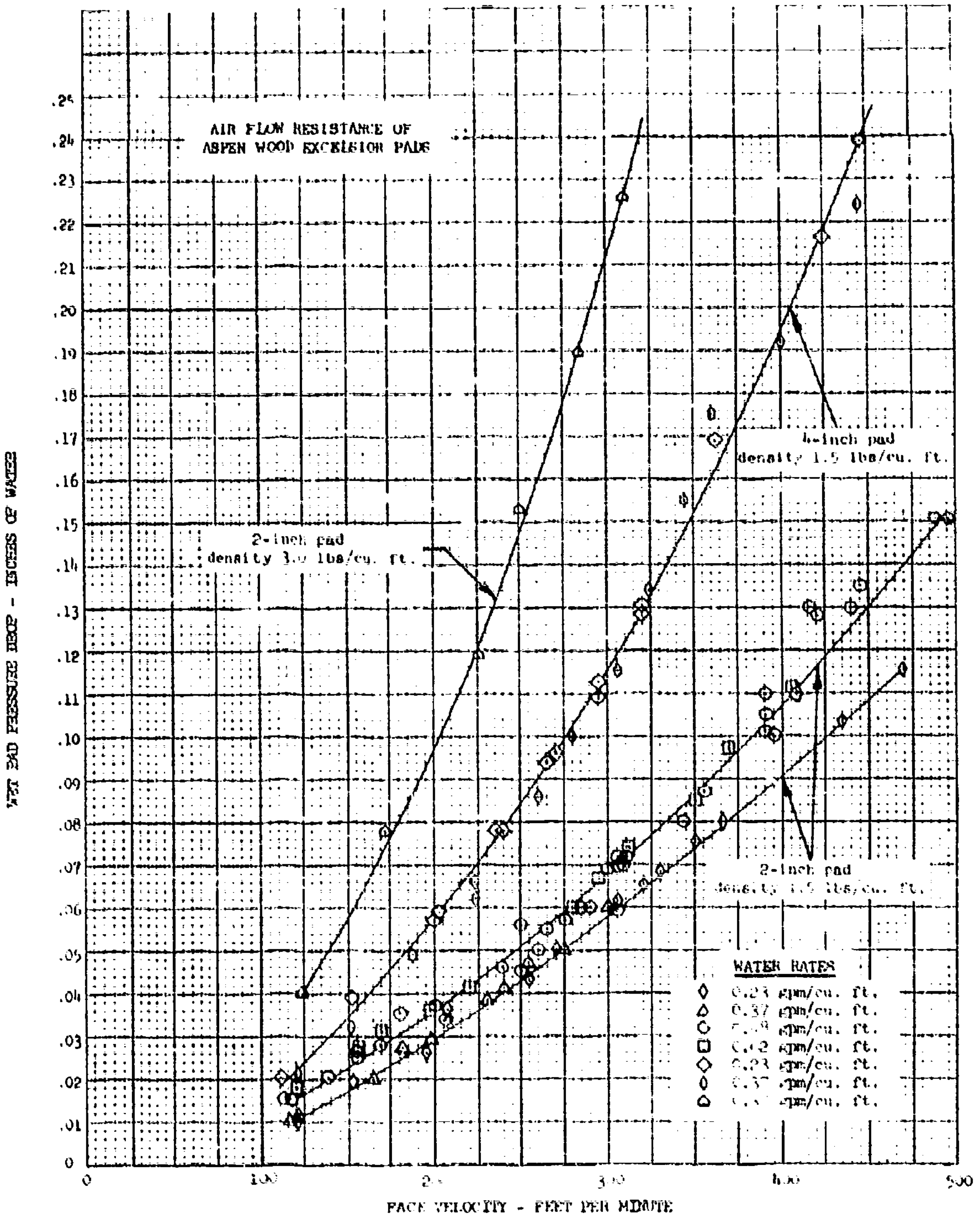


Figure 13 AIR FLOW RESISTANCE OF ASPEN WOOD EXCELSIOR PADS

Although the saturating effectiveness is increased considerably by doubling the mass density of the pad, approximately the same effectiveness (see Figure 14, p.30) can be obtained with a 4-inch thick pad at a reduced pressure drop of 0.116 iwg.

4.3 Error Analysis

The following error analysis is made in order to evaluate the accuracy of the saturating effectiveness as determined by equation 13, page 8.

Given a function of n number of variables, the relative error in P, $\frac{d(P)}{P}$, is defined as

$$\frac{d(P)}{P} = \frac{1}{P}\left[\left(\frac{\partial P}{\partial X_1}\right) d(X_1) + \left(\frac{\partial P}{\partial X_2}\right) d(X_2) + \ldots\ldots + \left(\frac{\partial P}{\partial X_n}\right) d(X_n) \right] \qquad (15)$$

where:

$d(P)$ = absolute error in the property, P

$d(X_1)$ = absolute error in the variable, X_1, due to experimental measurement

$\frac{\partial P}{\partial X_1}$ = absolute error contribution to the property, P, due to the variable, X_1, with all other variables held fixed.

Applying the above discussion to equation 13, the relative error in determining the saturating effectiveness is:

$$\frac{d(E)}{E} = \frac{1}{E}\left[\frac{X_2}{X_1} d(X_1) - \frac{1}{X_1} d(X_2) \right] \qquad (16)$$

where:

$X_1 = DB_1 - WB_1$ = inlet temperature depression

$X_2 = DB_2 - WB_2$ = outlet temperature depression.

Substituting for E in equation 13, the appropriate relation in terms of X_1 and X_2, the relative error then becomes

$$\frac{d(E)}{E} = \frac{1}{X_1 - X_2}\left[\frac{X_2}{X_1} d(X_1) - d(X_2) \right] \qquad (17)$$

The dry-bulb and wet-bulb temperatures were measured to the nearest half-degree, thus $d(DB) = d(WB) = \pm 1/4°F$ or $d(X_1) = d(X_2) = \pm 1/2°F$. During each test, the following relationships held:

$$10 \leq X_1 \leq 16$$

$$1/4 \leq X_2 \leq 2$$

Four combinations of X_1 and X_2 can be used to calculate the relative error using equation 17. The combination of $X_1 = 10$ and $X_2 = 1/4$, however, gives the maximum possible relative error, which is

$$\frac{d(E)}{E} \leq \pm\left[0.053\right]$$

$$d(E) \leq \pm\left[0.053\ E\right]$$

or the maximum possible relative error for any individual test is less than or equal to ± 5.3 percent of the experimentally determined saturating effectiveness.

4.4 Conclusions and Application

The saturating effectiveness and pad pressure drop for the tests conducted are summarized in Figure 14. Both doubling the thickness and doubling the density of a 2-inch thick pad with a density of 1.5 pounds of aspen wood per cubic foot of cell volume(lbs/cu ft) resulted in a large increase in the saturating effectiveness. Doubling the density, however, resulted in a pad pressure drop that was more than three times that of a 2-inch,1.5 lbs/cu ft density pad, while doubling the thickness resulted in a pressure drop which was only twice that of the 2-inch pad. Since the pressure drop of the 2-inch, 3.0 lbs/cu ft density pad is excessive, it is recommended that the use of the data be restricted to either a 2- or 4-inch pad with a density of 1.5 lbs/cu ft.

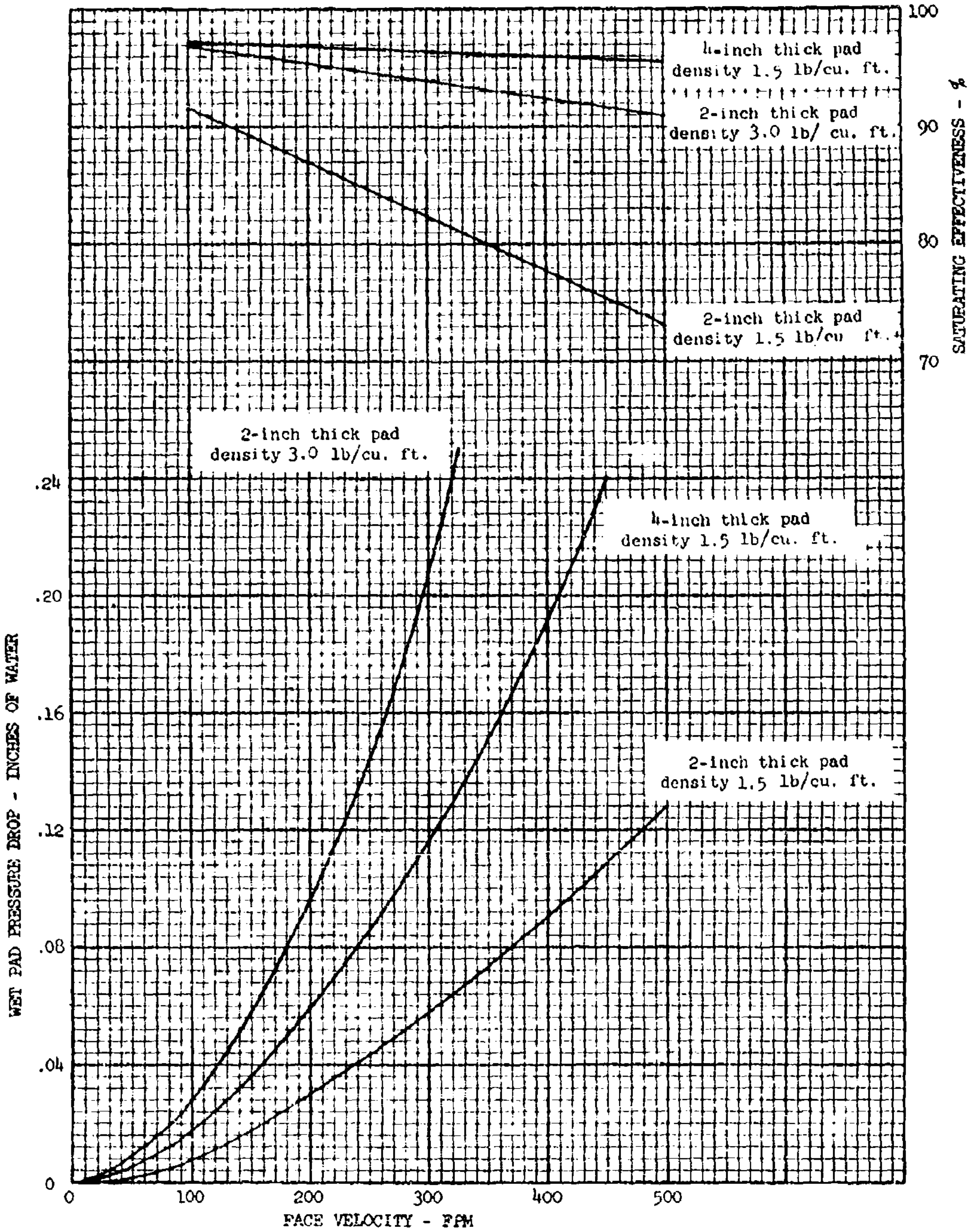


Figure 14 PERFORMANCE OF ASPEN WOOD EXCELSIOR EVAPORATING PADS

The design of a drip-type unit should be limited to face velocities up to 450 feet per minute, since above this face velocity, water carry-over occurs. The inlet air dry and wet-bulb temperature does not significantly affect the saturating effectiveness; however, the quantity of water which is evaporated is a function of the inlet air temperature. The quantity of water recirculated through the drip-type cooler must exceed five times the quantity being evaporated if the saturating effectivenesses shown in Figure 14 are to be achieved.

Before a decision can be made as to the merits of drip-type evaporative coolers for disposing of excess heat and humidity in protective shelters, a parametric and equipment systems analysis should not only evaluate the application of ventilators, refrigerating devices and drip-type evaporative coolers, but should also include the application of indirect (Ref. 7) and two-stage (Ref. 8) evaporative cooling. Use of evaporative cooling in shelters will result in reduced ventilation requirements as shown in a study by GARD (Ref. 9). Use of an indirect evaporative cooler also will give an air reduction; however, the relative humidity of the supply air can be reduced from values of greater than 90 percent to values of from 50 to 60 percent. Use of two-stage evaporative cooling results in a still further reduction of the ventilation requirements.

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SUMMARY
OF
RESEARCH REPORT

PERFORMANCE OF ASPEN WOOD EXCELSIOR FOR USE IN EVAPORATIVE COOLERS

OCD Work Unit 1214A

GARD Final Report 1268-1   January 1967

by

General American Transportation Corporation
General American Research Division
Thermal and Environmental Systems Department
Niles, Illinois

REVIEW NOTICE

This is a summary of a report which has been reviewed in the Office of Civil Defense and approved for publication. Approval does not signify that the contents necessarily reflect the views and policies of the Office of Civil Defense.

INTRODUCTION

Evaporative coolers are devices that have potential application in shelter ventilation and cooling equipment systems analysis. Commercially available evaporative coolers are the drip-type, spray-type, and the rotary pad-type (see Fig. 1). The drip-type evaporative cooler is the most common, and the pads are usually made of aspen wood excelsior. According to the International Metal Products Division of the McGraw-Edison Company (Ref. 1), the commercial units are designed and marketed with 2-inch thick aspen wood pads which have a density of 0.3 pound per square foot. The air system is nominally rated at a pad face velocity of 300 feet per minute and a water recirculation rate ten times that being evaporated, thus resulting in a saturating effectiveness of 80 percent. The resistance to air flow for these units is a nominal 0.1 inch water gage.

Since little performance data is available for aspen wood excelsior, this investigation was initiated because it appeared that the resistance to air flow by aspen wood is considerably less than that for any other medium. In addition, aspen wood excelsior is the least expensive medium available for use in evaporative coolers.

Ref.1 - Henninger, R. H., Private Communication with R. S. Ash, September, 1966.

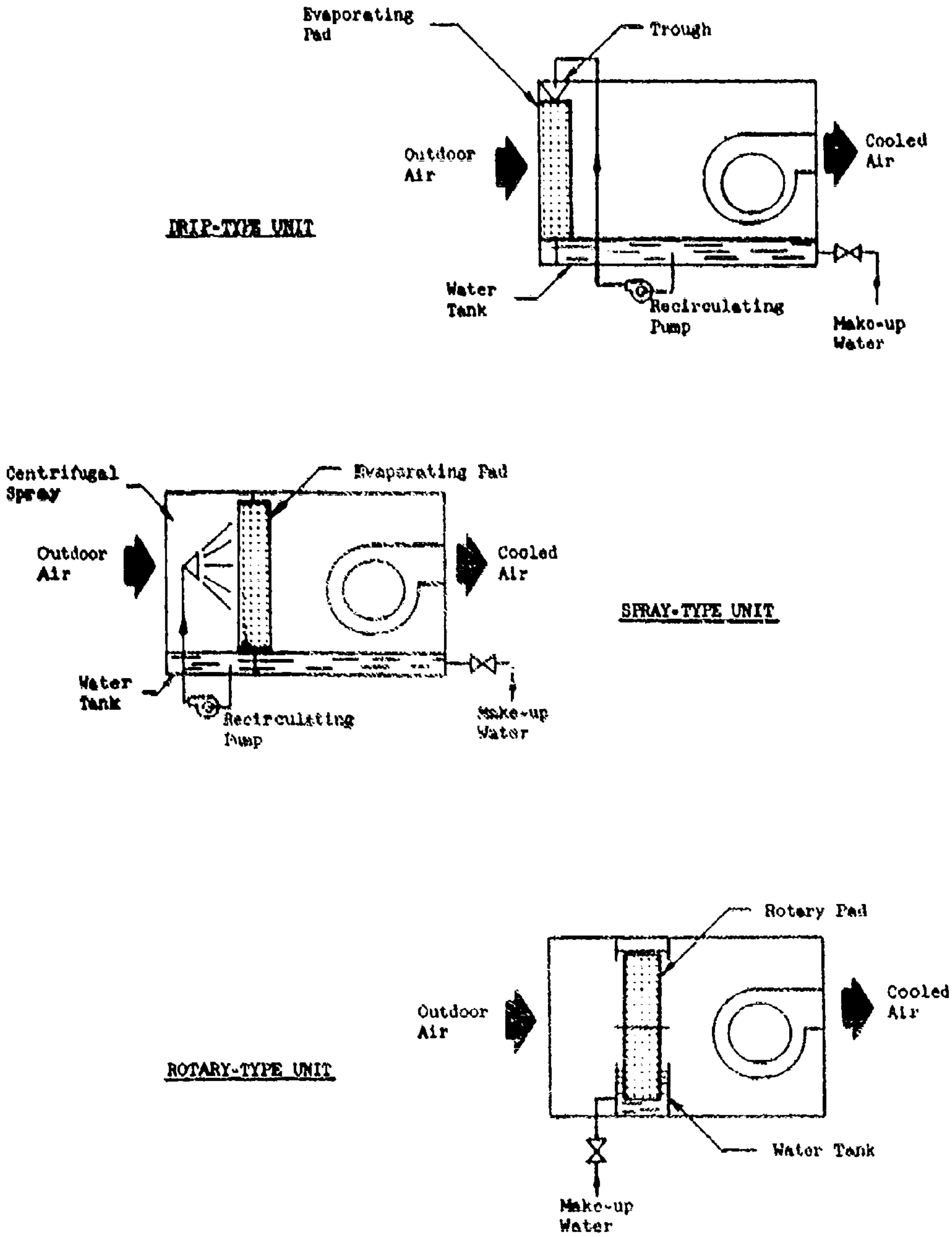

Figure 1 UNIT-TYPE EVAPORATIVE COOLERS

TEST APPARATUS AND PROCEDURES

The apparatus consisted of a test vehicle, a 30-foot air flow measuring station, and a test chamber as shown in Figure 2. With this arrangement, the face velocity through the evaporating pads, the inlet air dry and wet-bulb temperatures, the density and thickness of the pads, and the water flow to the pads could be controlled and varied.

Test Vehicle -- The air supply for the test chamber was obtained from the Office of Civil Defense (OCD) Test Vehicle No. 1. The Test Vehicle is capable of supplying up to 8,600 cfm of air at a predetermined dew-point temperature and a dry-bulb temperature.

Air Flow Measuring Station -- The design of the air flow measuring station was based on recommendations of the National Electrical Manufacturers Association, and was fabricated with rigid 12-inch diameter spiral conduit duct. Air flow rates were measured with an 8-inch diameter aperture, sharp-edge, orifice plate across which the differential static pressure was measured with a manometer connected to piezometer rings as shown in Figure 2. The orifice plate was calibrated with a pitot tube which was located four feet upstream from the orifice. The barometric pressure and dry and wet-bulb temperatures were recorded before each test to provide for correction of the air flow to standard air (0.075 pound per cubic foot).

Test Chamber -- The test chamber measured 4 feet by 4 feet by 2 feet and was constructed of 3/4-inch plywood to minimize heat exchange with the surroundings. The chamber housed the water distribution system, pads, and temperature measuring equipment. Air entered the chamber through a 12-inch diameter duct. To ensure that the face velocity entering the pad was uniform, the air was passed through a fine mesh screen. The variation in in face velocity entering the pad did not exceed ± 15 percent.

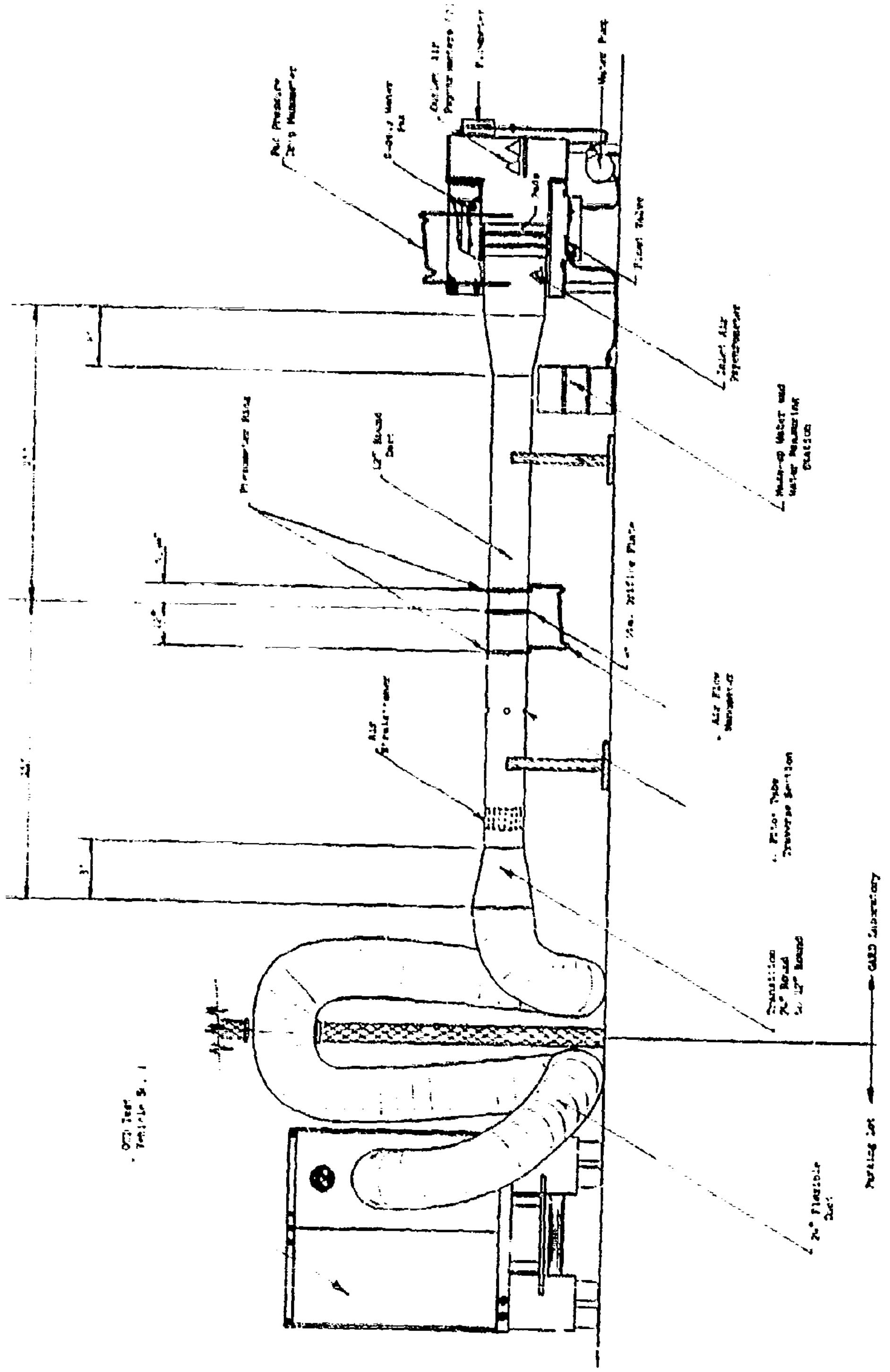

Figure 2 SCHEMATIC LAYOUT OF TEST APPARATUS

TEST RESULTS AND CONCLUSIONS

The saturating effectiveness and pad pressure drop for the tests conducted are summarized in Figure 3. Both doubling the thickness and doubling the density of a 2-inch thick pad with a density of 1.5 pounds of aspen wood per cubic foot of cell volume (lbs/cu ft) resulted in a large increase in the saturating effectiveness. Doubling the density, however, resulted in a pad pressure drop that was more than three times that of a 2-inch, 1.5 lbs/cu ft density pad, while doubling the thickness resulted in a pressure drop which was only twice that of the 2-inch pad. Since the pressure drop of the 2-inch, 3.0 lbs/cu ft density pad is excessive, it is recommended that the use of the data be restricted to either a 2- or 4-inch pad with a density of 1.5 lbs/cu ft. The design of a drip-type unit should [illegible] limited to face velocities up to 450 feet per minute, since above this face velocity, water carry-over occurs. The inlet air dry and wet-bulb temperature does not significantly affect the saturating effectiveness; however, the quantity of water which is evaporated is function of the inlet air temperature. The quantity of water recirculated through the drip-type cooler must exceed five times the quantity being evaporated if the saturating effectivenesses shown in Figure 3 are to be achieved.

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Ref. 2 - Baschiere, R. J., et. al., "Shelter Forced Ventilation Requirements Using Unconditioned Air", prepared for the Office of Civil Defense under Stanford Research Institute Subcontract No. B-[illegible]421([illegible]949A-4)-US, General American Transportation Corporation (GARD Report 1[illegible]), Niles, Illinois, February, 19[illegible].

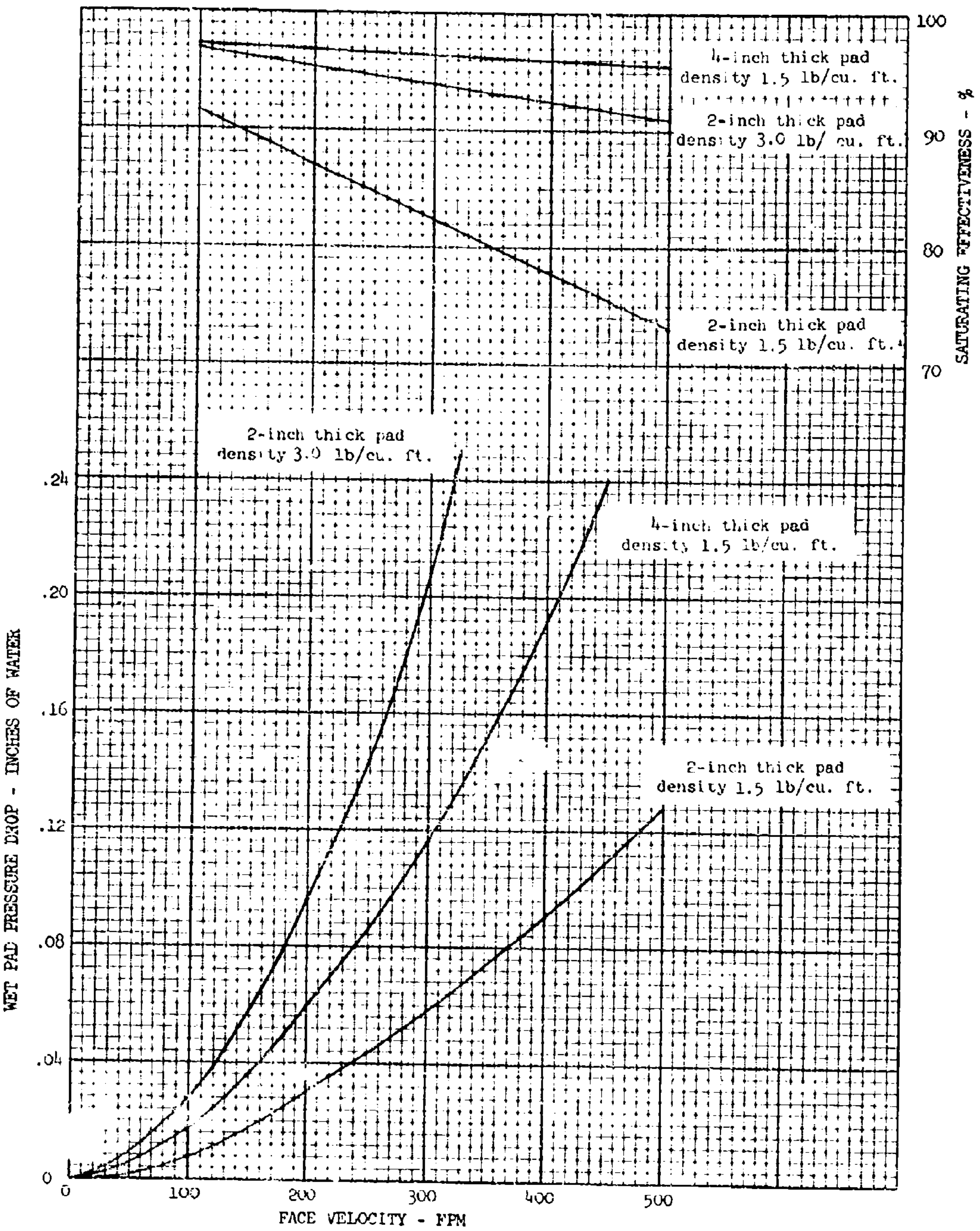

Figure 3 PERFORMANCE OF ASPEN WOOD EXCELSIOR EVAPORATING PADS

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13. ABSTRACT

Evaporative coolers are devices which have potential application in shelter ventilation and cooling equipment systems analysis. Reported herein are the results of the laboratory tests conducted to determine the saturating effectiveness and air flow resistance of aspen wood excelsior when used as the water evaporating media in a drip-type evaporative air cooler. The tests were conducted with a cell area of 4 sq. ft., 2 and 4" thick, media density of 1.5 and 3.0 lb/cu. ft. of cell volume, face velocities ranging from 100 to 500 ft/min., water flow rates from 0.23 to 0.62 gal./min., and inlet air conditions of [illegible]°F dry-bulb temperature (DBT) and 70°F wet-bulb temperature (WBT), 95°F DBT and 76°F WBT, and 90°F DBT and 75°F WBT. The saturating effectiveness for the 2-in. thick pad of 1.5 lb/cu. ft. of cell volume decreased steadily from 92% at 100 ft/min face velocity to 73% at 500 ft/min. Doubling the thickness resulted in a saturating effectiveness that was almost independent of the face velocity and constant at 97%, while doubling the media mass density gave a saturating effectiveness that varied from 97% to 100 ft/min. to 91% at 500 ft/min. To obtain these saturation effectivenesses, the quantity of water recirculated in the drip-type cooler must exceed 5 times the amount evaporated into the air. The air flow resistance of the 2-in. thick pad ranged from 0.017 in. of water at 150 ft/min. to 0.09 in. of water at 400 ft/min. Doubling the thickness of the pad resulted in an air flow resistance which was twice that of the 2-in. pad, while doubling the density increased the air flow resistance by more than three times that of the 2-in. pad. (U)

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